BEST AVAILABLE COPY

(19) Canadian Intellectual Property Office — An Agency of Industry Canada
Office de la Propriété Intellectuelle du Canada — Un organisme d'Industrie Canada

(11) CA 2 264 497 (13) A1
(40) 02.09.2000
(43) 02.09.2000

(12)
(21) 2 264 497
(51) Int. Cl.^{6}: F16K 015/04
(22) 02.03.1999

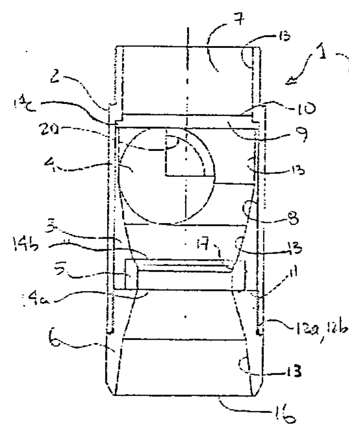
(71)
CONN, KENNETH S.,
29 Lang Crescent S.W., MEDICINE HAT, A1 (CA).

(72)
CONN, Kenneth S. (CA).

(74)
GOODWIN, SEAN W.

(54) LANTERNE ET BILLE DE SOUPAPE POUR POMPE ALTERNATIVE
(54) VALVE CAGE AND BALL FOR A RECIPROCATING PUMP

(57)
An improved ball-type check valve is provided for a reciprocating pump. The valve is an assembly having a cage containing the ball and a ball stop at the cage's fluid outlet, a ball seat and a seat retainer. A contiguous bore is formed therethrough from a fluid inlet at the retainer to the fluid outlet. The bore has an inside wall which is formed of three continuous profiles. The diameter of the bore diminishes from the fluid inlet of the seat retainer to the ball seat, then increases in the cage to the fluid outlet. Bore transitions are substantially tangential throughout. The cross-sectional area of the cage is managed to be about twice that of the ball's area. The bore of the ball seat is maximized to be within 1/8 inch of the diameter of the ball. More preferably, the ball is hollow. Accordingly, the improved valve has a low pressure drop and can operate at lower hydraulic head before vapor-locking or reduced pumping efficiency. 19



OPIC
OFFICE DE LA PROPRIÉTÉ INTELLECTUELLE DU CANADA

CIPO
CANADIAN INTELLECTUAL PROPERTY OFFICE

(12) (19) (CA) **Demande-Application**

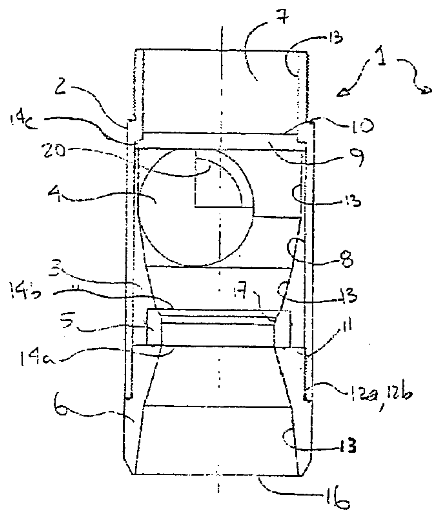
(21) (A1) **2,264,497**
(22) 1999/03/02
(43) 2000/09/02

(72) CONN, Kenneth S., CA
(71) CONN, Kenneth S., CA

(51) Int.Cl.^{6} F16K 15/04

(54) **LANTERNE ET BILLE DE SOUPAPE POUR POMPE ALTERNATIVE**

(54) **VALVE CAGE AND BALL FOR A RECIPROCATING PUMP**



(57) An improved ball-type check valve is provided for a reciprocating pump. The valve is an assembly having a cage containing the ball and a ball stop at the cage's fluid outlet, a ball seat and a seat retainer. A contiguous bore is formed therethrough from a fluid inlet at the retainer to the fluid outlet. The bore has an inside wall which is formed of three continuous profiles. The diameter of the bore diminishes from the fluid inlet of the seat retainer to the ball seat, then increases in the cage to the fluid outlet. Bore transitions are substantially tangential throughout. The cross-sectional area of the cage is managed to be about twice that of the ball's area. The bore of the ball seat is maximized to be within 1/8 inch of the diameter of the ball. More preferably, the ball is hollow. Accordingly, the improved valve has a low pressure drop and can operate at lower hydraulic head before vapor-locking or reduced pumping efficiency.

Industrie Canada Industry Canada

ABSTRACT OF THE INVENTION

An improved ball-type check valve is provided for a reciprocating pump. The valve is an assembly having a cage containing the ball and a ball stop at the cage's fluid outlet, a ball seat and a seat retainer. A contiguous bore is formed therethrough from a fluid inlet at the retainer to the fluid outlet. The bore has an inside wall which is formed of three continuous profiles. The diameter of the bore diminishes from the fluid inlet of the seat retainer to the ball seat, then increases in the cage to the fluid outlet. Bore transitions are substantially tangential throughout. The cross-sectional area of the cage is managed to be about twice that of the ball's area. The bore of the ball seat is maximized to be within 1/8 inch of the diameter of the ball. More preferably, the ball is hollow. Accordingly, the improved valve has a low pressure drop and can operate at lower hydraulic head before vapor-locking or reduced pumping efficiency.

"VALVE CAGE AND BALL FOR A RECIPROCATING PUMP"

FIELD OF THE INVENTION

This invention relates to improvements to a valve, ball seat and ball used in a subterranean reciprocating pump.

BACKGROUND OF THE INVENTION

A reciprocating, positive displacement pump is typically used for pumping subterranean fluids containing fine solids to surface. The pump comprises a cylindrical pump barrel, piston and piston rod. The bore of the pump barrel forms a compression chamber. The piston rod reciprocates up-and-down in the compression chamber to complete one pumping cycle. The piston rod is suspended from a rod string or reciprocating production tubing. The barrel is anchored in the casing of a well. A standing valve is located at the bottom of the barrel. A piston and travelling valve are located at the bottom of the piston rod. To begin a pumping cycle, on the upstroke of the piston rod, the standing valve opens in and permits fluid to fill the compression chamber of the pump barrel. On the downstroke of the piston rod, the standing valve closes and the travelling valve opens to permit compressed fluid in the barrel to enter the piston rod. This completes the pumping cycle. During the next up stroke of the piston rod, the fluid within the rod is incrementally advanced up the well, while new fluid is drawn into the compression chamber.

The standing and travelling valves are typically identical and comprise a form of ball check valve. More particularly, having reference to

prior art valve shown in Figs. 1,1a, the valve comprises a guide race or cage. The cage is generally tubular in configuration and defines an axial bore extending therethrough. A ball is fitted within the bore of the cage. Adjacent the cage's lower end is positioned a ball seat ring. The seat ring serves two functions: one, to retain the ball within the bore; and two, to form a seal with the ball. Adjacent the cage's upper end is a horizontal bar or stop extending across the bore for preventing exit of the ball.

With the advent of horizontal wells, the bore of more contemporary valve cages include longitudinal guide ribs (Fig. 1a) which form a race of substantially constant diameter for closely guiding the ball so that only moves axially. Presumably this is to aid in guiding the ball directly and concentrically onto the race during the closing portion of the cycle. Fluid flows between the ribs and around the ball.

Typically, the ball seat is sandwiched between the cage and a lock ring or seat retainer screwed into the bottom of the cage. As shown in Fig. 1, there is a sharp or abrupt transition between the bores of the lock ring, the ball seat ring and the guide race of the cage.

In Alberta, Canada, valves must contend with difficult environments. More particularly, pumping scenarios include:

- cold production of heavy oils containing large fractions of sand and potentially sandstone solids;
- Steam injected and other thermal wells in which contained water tends to flash as pressure is reduced; or
- slant or horizontal wells in which the valves are required to perform on their sides.

Application of conventional valves in the above scenarios often result in low efficiencies. Conventional wisdom in this art specifies that the diameter of the ball should be at least ½ inch larger than the diameter of the ball seat ring so as to avoid sticking or jamming of the ball.

Unfortunately, the resulting abrupt transition for fluid entering the small diameter of the ball seat, and leaving it, results in high pressure drop. In cold production of heavy oil, this severely limits the intake of virgin oil to the pump barrel. In thermal cases, the pressure drops usually causes contained water to flash to steam, causing a 1000-fold increase in volume, displacing oil and vapor-locking the pump or at least reducing pumping efficiencies.

To combat the low efficiencies associated with thermal applications, operators sometimes compensate by landing the pump deeper in the well, often lower than the heel or deep in the horizontal portion of horizontal well, thereby increasing the hydrostatic head or pressure. There is significant cost associated with such a remedy. Landing a pump deeper in a well requires more time, greater quantities of production tubing, and requires actuation of the pump around the heel, resulting in high cyclical stresses in the rod or production string. Wear associated with cyclical movement through the heel has prompted implementation of expensive tubing and rod rotation devices.

Again, in cold production scenarios, large particle debris such as sandstone often become lodged between the ball, the cage's bore, and the guide ribs.

Ideally, pressure drop should be minimized, the flow passages through the valve should be maximized, and should be achieved without compromising valve integrity. It is known that ball-type check valves are associated with a pressure drop equivalent to that experienced in about 150 pipe diameters.

In some prior art implementations, efforts have been made to utilize the largest possible valve cage and ball combination. Unfortunately, as the size of the valve increases, the ball becomes so heavy that excessive pressure drops are incurred merely to lift the ball. A 3.75 inch diameter ball weighs almost 8 pounds (3.5 kg). Some prior art remedies include manufacture of the ball of lightweight and very expensive titanium – being only about 56% the weight of steel alloys.

SUMMARY OF THE INVENTION

Generally, the objectives of minimizing pressure drop through the valve while maximizing flow passage therethrough are met by improving the conventional valve in the following manner:

- managing the profile of the bore through the valve from its inlet through its exit for avoiding abrupt transitions and the associated entrance and exit losses, more particularly by forming a continuous and tangential inside wall throughout;
- maximizing the diameter of bore of the ball seat;
- minimizing turbulence through the fluid bore and around the ball;
- eliminating guide ribs for enabling lateral movement of the ball, thereby creating the largest possible particle passing bore size; and
- minimizing the weight and inertia of ball.

By implementing the above factors, the improved valve demonstrates increased pump efficiency and thereby has a reduced need for the pump to be landed so deep in a well. In horizontal wells, this translates into the ability to land the pump higher in the well, more specifically in the vertical portion of the well above the heel, and thereby receive all of the benefits associated thereof. Additionally, in vertical wells, the pump can be repositioned above the perforations and avoid sanding and debris issues.

In a broad aspect, an improved valve is provided for a reciprocating pump. The valve comprises a cage containing a ball and ball stop at a fluid inlet, a ball seat and a seat retainer. A contiguous bore is

formed therethrough. The bore has an inside wall which is formed of three continuous profiles. The diameter of the bore diminishes from the fluid inlet of the seat retainer to the ball seat, then increases in the cage to the fluid outlet. Bore transitions are managed so as to be substantially tangential throughout.

Preferably, the cross-sectional area of the cage is about twice that of the cross-section area of the ball. Further, it is preferable to maximize the bore of the ball seat to be within 1/8 inch of the diameter of the ball. Even more preferably, the ball is manufactures so as to be hollow, thereby reducing its inertia.

A BRIEF DESCRIPTION OF THE DRAWINGS

Fig. 1 is a cross-sectional side view of a valve according to the prior art;

Fig. 1a is a cross-sectional view of the valves of Fig. 1 cut through the valve along lines I-I;

Fig. 2 is an exploded, cross-sectional view of a valve according to one embodiment of the invention.

Fig. 3 is an assembled, cross-sectional view of the valve of Fig. 2, shown in operation wherein fluid is passing past the ball. The ball is shown to having one quadrant partially cut away to illustrate the optional hollow ball embodiment; and

Fig. 4 is an assembled, cross-sectional view of the valve of Fig. 2, shown in operation wherein the ball is seated and flow is blocked. The left cross section shows a stepwise inside wall transition and the right cross-section depicts a smoother continuous profile.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT

As shown in the exploded view of Fig. 2, one embodiment of the valve of the present invention is a valve assembly 1 comprising a cylindrical valve body 2, a valve guide insert or cage 3, a ball 4, a ball seat 5, and a seat retainer 6.

Throughout, despite the potential for orientations of the valve 1 in horizontal well applications, terms such as top, upward and their counterparts relate to orientations of the valve as if it was installed in a vertical implementation.

As shown in the assembled views of Figs. 3 and 4, a fluid bore 7 is formed through the valve assembly, and listed in order of the direction of fluid flow, is continuous through each of the seat retainer 6, the seat 5, the cage 3, and the body 2.

The valve body 2 forms an assembly bore 8 for accepting the valve guide insert or cage 3. The assembly bore can accept a variety of cages dependent upon the application.

The ball 4 is located within and is movable within the cage's fluid bore 7. A stop pin 10 extends across the top of the cage 3 to prevent upward escape of the ball 4. The ball seat 5 is fitted at the bottom of the cage to prevent downward escape of the ball 4. The ball seat 5 and cage 3 are retained within the valve body 2 by the seat retainer 6. The outer circumference of the seat retainer 6 is threaded 12a for complementary engagement with an inner threaded circumference 12b of the lower end of the valve body 2.

The exterior of the valve body shown in Figs. 2 – 4, and the means for affixing the valve body to the pump, are shown as configured for a standing valve. The exterior of a valve body, and means for affixing the valve as a traveling valve, is not shown however, the interior of the valve 1, and more particularly the fluid bore 7, are the same whether implemented as a standing, or a traveling valve.

The fluid bore 7 comprises an inside wall 13. The interfaces 14a-14c of the wall 13 between each component are continuous, namely: the interface 14a between the seat retainer 6 and the ball seat 5; the interface 14b between the ball seat 5 and the cage 3; and possibly an

interface 14c between the cage 3 and the valve body 2. Very simply, this is accomplished by removing any intermediate and abrupt transitions and thereby minimizing associated loss of head (increased pressure drop). More specifically, the diameter of the fluid bore 7 at the interfaces 14a,14b,14c are matched and the transition is made as close to tangent as possible.

As shown in Figs. 3 & 4, machining techniques result in step transitions S1,S2,S3 on the profile of the inside wall 13. Referring to Fig. 4, the left cross-section shows the step transitions S1,S2,S3. The right cross-section shows two continuous transitions S4,S6 which is accomplished using numerical machining techniques.

Accordingly, there is minimal formation of low-pressure areas, or more particularly, there is minimization of entrance and exit loss pressure drop which results from the formation of vena-contracta flow. The profiles of the inside wall between each of the seat retainer, the ball seat, the cage, and the valve body are continuous and substantially tangent.

More particularly, the fluid bore 7 in the seat retainer 6 forms a first profile P1 or bell-like fluid intake which tapers inwardly from the bottom 16 to the ball seat 5 (diminishing diameter). The diameter of the fluid bore 7 of the seat retainer 6 at the interface 14a matches that of the ball seat 5 and is substantially tangent thereto.

The ball seat 5 comprises a circumferential seal 17, typically manufactured of a stellite alloy. The inside wall 13 of the ball seat 5 forms a second profile P2 having an outwardly tapering discharge (increasing diameter). The diameter of the seat's bore 7 and circumferential seal 17 is maximized. In contradistinction to the practice of the prior art, the difference

between the diameter of the ball 4 and the diameter of the bore 7 of the ball seat 5, is less than the prior art ½ inch, most preferably in the order of 1/8 inch on the diameter. For example, the bore 7 of the ball seat 5 for a 3.75 inch ball has a diameter of 3.625 inches. It is clear upon examination that the energy loss or pressure drop though the ball seat 5 and past the ball 4 is significantly reduced (compare Fig. 1 and Fig. 3) as the flow is no longer subjected to such an abrupt transition at interface 14b, nor is it deflected laterally as much as it is in prior art valves.

The circumferential seal 17 is beveled steeply (only 11 degrees off the axis) to match the deep-seating of the ball 4 in the ball seat 5.

The diameter of the second profile P2 at the discharge of the ball seat 5 matches the fluid bore 7 at the inlet of the cage 3 and is substantially tangent thereto.

The inside wall 13 of the cage 3 forms a third profile P3 which has a progressively diminishing side wall thickness between the ball seat 5 and stop pin 10 (increasing diameter). Accordingly the diameter of the fluid bore 7 increases upwardly towards the stop pin 10. This provides maximal flow area and minimizes pressure drop.

Preferably, the bore 7 of the ball seat 5 is as large as possible, but the corresponding ball 4 should not be made too large for the fluid bore 7 in the cage.

More specifically, the diameter of the ball 4 is restricted such that the cross-sectional area of the fluid bore 7 around the ball during fluid flow is about equal to or greater than two times the ball's cross-sectional

area. Note that the ball rises in the cage during fluid flow to rest against the stop pin 9.

First then, the fluid bore 7 is made as large as it can be. Then the ball's size is chosen so that the projected area of the ball 4 is about ½ of the cross-sectional area of the bore 7 of the cage 3 when the ball 4 is against the stop pin 9. The size of the ball seat 5 is then chosen so that the diameter of the bore 7 through the ball seat 5 is only about 1/8" less that the diameter of the ball 4. Preferably, the area of the bore through the ball seat is comparable to the free area in the cage's bore 7 around the ball 4.

The stop pin 9 itself comprises a slender member extending across the center of the fluid bore, blocking only enough of the fluid bore to prevent egress of the ball. Accordingly, as shown in Fig. 3, the distance between the inside wall and the stop pin is greater that the radius of the ball so that the unseated ball will rest either one side of the pin or the other. As a result, lateral ball movement is arrested and the ball is unable to spin or flutter during high fluid flow, avoiding undesirable turbulence, pressure drop and associated gas breakout.

The relationship between the diameter of the ball 4, the increasing diameter fluid bore 7 in the cage 3, and the distance between the ball seat 5 and stop pin 9 is designed to minimize the axial travel of the ball 4 and thereby provide the fastest ball response time for engaging and disengaging the ball seat (closing and opening respectively).

Specifically, no prior art guide ribs are provided, thereby permitting the ball to move laterally within the cage and thereby expose or form the largest possible flow passage through the cage 3, improving fluid

flow with the lowest associated pressure drop, and for passing the largest possible entrained particles. As a result, screening of the valve inlet and accumulation of debris (particles) is avoided.

In a second embodiment, the normally solid ball 4 is manufactured in a hollow form. By reducing the weight of the ball, less pressure drop is incurred in lifting the ball from its seat and thereby further reduces the opportunity for gas breakout or flashing to occur. The reduced inertia of the ball also improves response of the ball to opening and closing.

While hollow balls are known in the prior art of steam traps, the known balls have insufficient wall thickness to survive in the environment of a subterranean reciprocating pump. Accordingly, novel manufacturing techniques are employed comprising machining of concave hemispherical hollows 20 in each of two matched metal blanks (not shown). Suitable materials of construction are 52100 chromium alloy or 440C stainless steel. The two blanks are welded together with the hollows 20 facing. The welded blanks are machined to form a sphere. The sphere is treated to homogenize, stress relieve and harden the sphere. Finally, the sphere is ground and surface hardened in known ball-bearing forming processes to form a perfectly sphere and the valve ball 4.

Example I

A test valve constructed according to the first embodiment was placed in a producing well in Northern Alberta to replace a prior art valve. The well was subjected to steam injection being a candidate for low efficiency pumping. Prior art valves had been used and were of the straight

bore, guide-rib type having 3.75" balls and 3.25" ball seats. These prior art valves required regular replacement on about a one month cycle. Specifically, the valve seats were "washing-out".

The test valve was formed with an inside wall profile as shown in Fig. 3. The maximum bore between the ball seat and fluid outlet had a diameter of about 5.25 inches for a cross sectional area of 21.64 square inches. A 3.75 inch ball was provided having a cross-sectional projected area of 11.04 inches. Accordingly the bore's cross-section was 1.96 times the ball's projected area so that as much free cross-sectional area remained through the bore as was blocked by the ball. For the 5.25 inch cage, use of a 4 inch ball would have been inappropriate, resulting in a ratio of bore-to-ball of only 1.76. This would have meant a further 14% decrease in the remaining free cross-sectional area in the bore due to the 4 inch ball.

A large diameter bore ball seat was used; being 3.625 inches in diameter or only 1/8 inch smaller in diameter than the ball. This seat provided a flow cross-sectional area of 10.32 square inches, or as much as 97% of the free cross-sectional around the ball.

As of the date of this application, the test valve has operated for 4 months without sticking or service of any kind, and continues to operate continuously, 24 hours/day with comparable pump efficiencies.

THE EMBODIMENTS OF THE INVENTION IN WHICH AN EXCLUSIVE PROPERTY OR PRIVILEGE IS CLAIMED ARE DEFINED AS FOLLOWS:

1. An improved valve assembly for a reciprocating pump, the assembly having a ball seat sandwiched between a cage and a seat retainer, the cage having a fluid outlet end and the seat retainer having a fluid inlet end and a contiguous fluid bore formed between the fluid inlet and outlet ends, a ball located within the cage for alternately engaging the ball seat to arrest fluid flow through the bore and disengaging from the seat to allow fluid flow through the bore, and a ball stop at the adjacent the fluid outlet to retain the ball within the cage, the improvement comprising:

a) an inside wall formed in the bore between the fluid inlet and outlet and having three profiles, the first profile formed between the fluid inlet and the ball seat in which the diameter of the inside wall diminishes to the ball seat;

b) a second profile formed in the bore of the ball seat in which the diameter of the inside wall of the ball seat is sufficiently small to prevent passage of the ball, and the transition between the first and second profile is substantially tangential; and

c) a third profile formed in the bore between the ball seat and the fluid outlet in which the diameter of the inside wall increases to the fluid outlet, the transition between the second and third profiles is substantially tangential, the resulting bore for the third profile having an obstruction-free

cross-sectional area in which the ball can move laterally and expose a large flow passage therethrough.

2. The improved valve assembly of claim 1 wherein the largest diameter bore of the cage has a cross-sectional area which is about twice the cross-sectional area of the ball.

3. The improved valve assembly of claim 2 wherein the difference between the diameter of the bore seat and ball is less than ½ inch.

4. The improved valve assembly of claim 3 wherein the difference between the diameter of the bore seat and ball is about 1/8 inch.

5. The improved valve assembly of claim 1 wherein the ball stop comprises a slender member extending across the fluid bore, the distance between the inside wall and the slender member being greater than the radius of the ball so that the ball, when disengaged from the ball seat, will rest either one side of the member or the other for arresting lateral ball movement.

6. The improved valve assembly of claim 5 wherein the ball is hollow.

7. The improved valve assembly of claim 1 wherein the ball is hollow.

8. The improved valve assembly of claim 7 wherein the hollow ball comprises first and second metal blanks having hemispherical hollow formed therein and welded together so that the hemispherical hollows form an internal spherical hollow, the exterior of the welded blanks being formed into a sphere with a significant wall thickness remaining between the exterior and the internal hollow.

9. The improved valve assembly of claim 2 wherein the ball is hollow.

10. The improved valve assembly of claim 3 wherein the ball is hollow.

11. The improved valve assembly of claim 4 wherein the ball is hollow.

12. A process for minimizing the pressure drop through a valve assembly for a reciprocating pump, the assembly having a ball seat sandwiched between a cage and a seat retainer, the cage having a fluid outlet end and the seat retainer having a fluid inlet end and a contiguous fluid bore formed between the fluid inlet and outlet ends, a ball located within the cage for alternately engaging the ball seat to arrest fluid flow through the bore and

disengaging from the seat to allow fluid flow through the bore, and a ball stop at the adjacent the fluid outlet to retain the ball within the cage, the process comprising:

a) providing a continuous inside wall in the bore between the fluid inlet and outlet, the inside wall having three profiles, the first profile formed between the fluid inlet and the ball seat, a second profile formed in the bore of the ball seat, and a third profile formed in the bore between the ball seat and the fluid outlet, the diameter of the inside wall of the ball seat being sufficiently small to prevent passage of the ball;

b) diminishing the diameter of the first profile between the fluid inlet and the ball seat;

c) increasing the diameter of the third profile between the ball seat and the fluid outlet; and

d) providing substantially tangential transitions between each of the first and second, and second and third profiles.

13. The process of claim 12 further comprising:

e) sizing the cross-sectional area of the third profile adjacent the fluid outlet to be about twice the cross-sectional area of the ball.

14. The process of claim 13 further comprising:

f) sizing the ball seat so that the difference between the diameter of the bore seat and ball is less than ½ inch.

15. The process of claim 14 further comprising:

f) sizing the ball seat so that the difference between the diameter of the bore seat and ball is about 1/8 inch.

16. The process of claim 15 further comprising:

g) forming the ball stop of a slender member extending across the fluid bore, the distance between the inside wall and the slender member being greater than the radius of the ball so that the ball, when disengaged from the ball seat, will rest either one side of the member or the other for arresting lateral ball movement.

17. The process of claim 16 further comprising providing a hollow ball.

18. The process of claim 12 further comprising providing a hollow ball.

Fig. 1
(Prior Art)

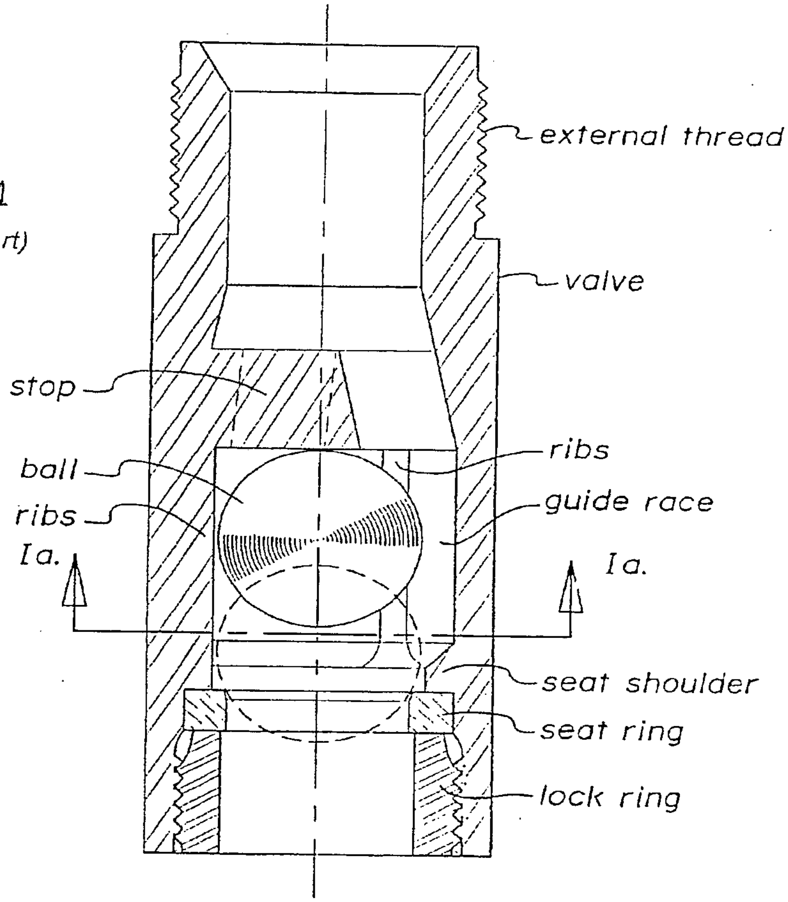


Fig. 1a.
(Prior Art)

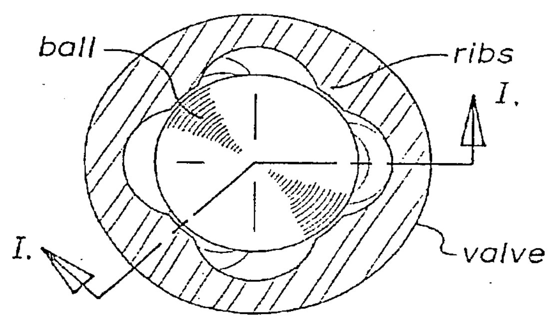
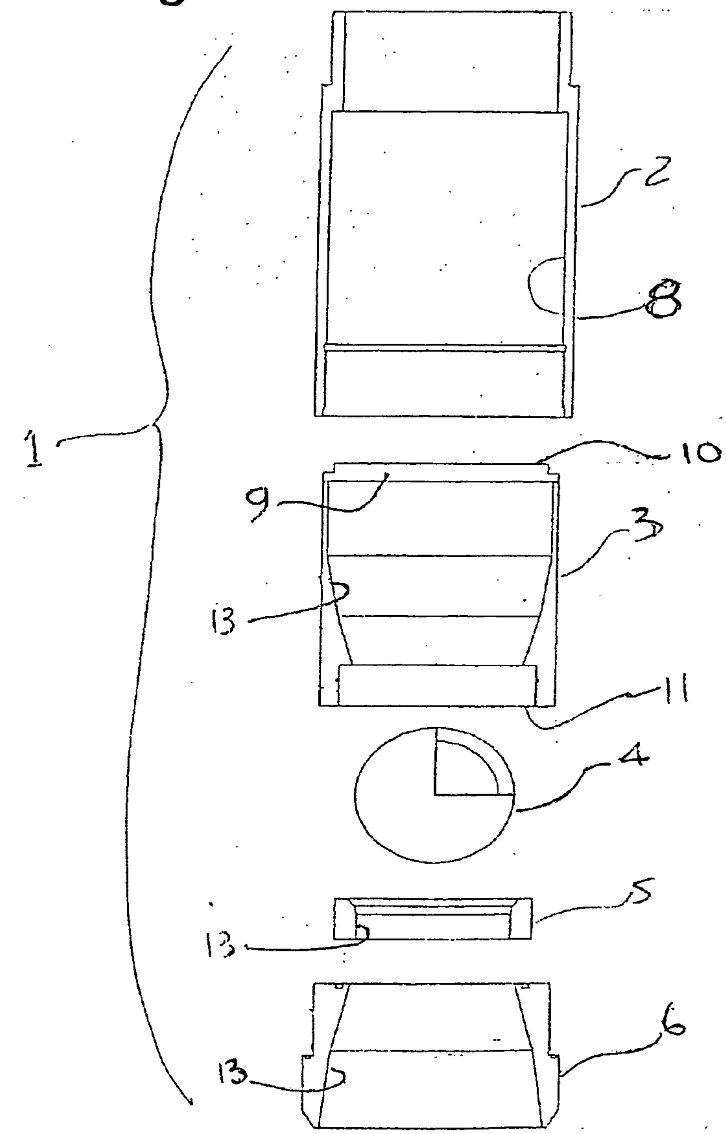


Fig. 2



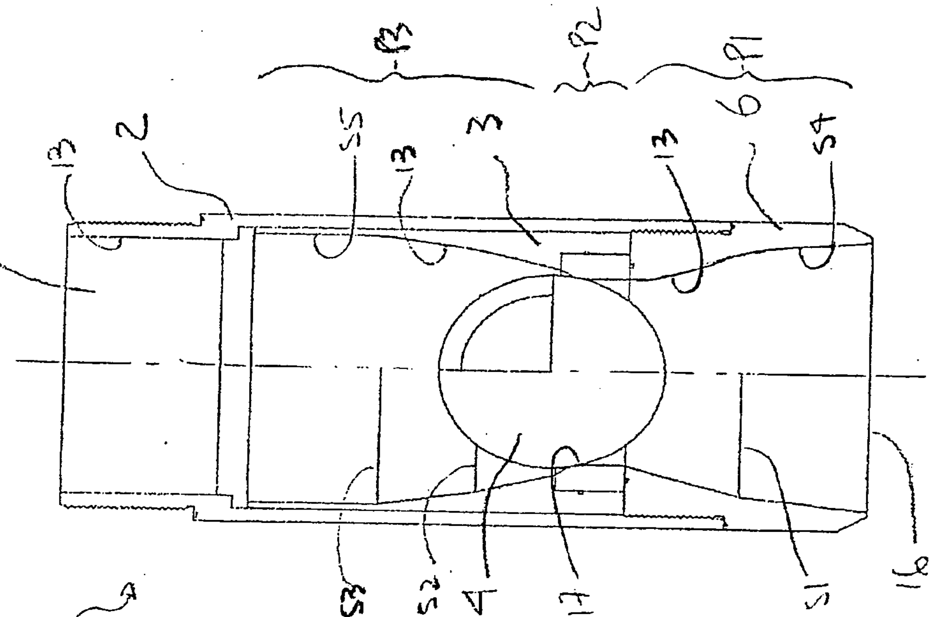

7
13
2
55
13
3
13
6
54
R3
R2
R1
53
52
4
17
51
16

Fig. 4

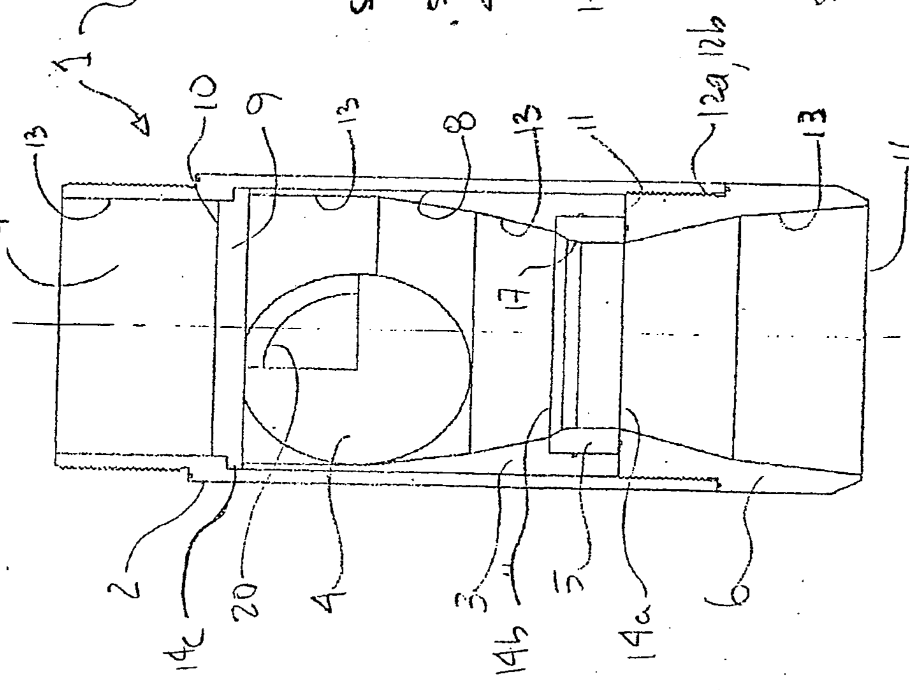

1
13
7
10
9
13
8
13
11
12a, 12b
13
16
2
14c
20
4
3
14b
5
14a
6
17

Fig. 3